API 613, FIFTH EDITION, SPECIAL PURPOSE GEAR UNITS FOR PETROLEUM, CHEMICAL AND GAS INDUSTRY SERVICES— OVERVIEW PRESENTATION

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ABSTRACT

Summary of Purpose for the Standard

API 613 provides a conservative basis for building critical service process industry turbomachinery gear unit drives. It is intended to provide gear units that give reliable trouble-free service when used in challenging operating and environmental conditions.

HISTORY

Before API 613 existed, gear unit drives for process industry applications were built to American Gear Manufacturers Association (AGMA) standards or standards of other countries. These standards established a design basis using analytical techniques with many factors that addressed the potential loads that could be applied and how those loads would be accommodated by the gear unit. Working with the gear manufacturers, the American Petroleum Institute (API) developed a simplified gear element rating formula that first appeared in API 613, Second Edition, in 1977. API 613, Second Edition, rating methods, based on AGMA rating formulas, established a conservative set of design parameters factored off of marine propulsion mechanical drive application practices for ships. This approach established a default design basis for gear element rating that removed the discretionary selection by gear designers of rating factors in the rating formulas of the applicable manufacturing standards such as AGMA. This same basic approach has been carried over to the latest edition of API 613 with each subsequent revision since API 613, Second Edition, based on the latest applicable AGMA standards. Since 1977, the gear units manufactured to API 613, have proven to be very reliable and have met user expectations consistent with availability requirements necessary for the industries in which they are applied.

REVIEW OF CONTENT

Scope

The Standard covers special purpose enclosed precision single and double helical one- and two-stage speed increasers and reducers of parallel shaft design for petroleum, chemical, and gas industry services. The Standard is primarily intended for gear units that are in continuous service applications without installed spare equipment.

Typical applications where these gear units are applied include:

- Centrifugal compressors and rotary positive displacement compressors.
- Blowers and fans.
- Centrifugal pumps.
- Reciprocating compressors.
- Extruders and mixers.
- Generators.

Gear Element Rating Methods

Gear element load ratings are based on tooth pitting index (K) and bending stress number (S) at rated power, times a service factor based on the application. Tooth pitting index corresponds to a contact surface stress number. It is used to determine a load rating at which progressive pitting of the gear teeth will not occur during their design life. Bending stress number is calculated at the tooth root filet and is designed to be below the maximum allowable stress possible without fatigue cracking. API 613, Fifth Edition, recommended service factors range from a low of 1.1 for a base load generator to as high as 2.3 for reciprocating pumps driven by reciprocating engines.

The API 613 calculations for tooth pitting index and bending stress are based on AGMA methods with a preselected set of design factors applied to ensure conservative rating under the load conditions for the application. A detailed comparison of API 613 versus AGMA 2101 (1996) rating factors and calculation methods is included in Appendix J of API 613 (attached for reference under APPENDIX A).

One way to present the difference between the size of the gear-set per API 613 and AGMA 6011 (1998) is to pictorially represent them. Illustrated in Figure 1 is a 20,000 horsepower reference unit with an API 613 gear-set on the left and an AGMA 6011 gear-set on the right, both with a service factor of 1.1 The AGMA 6011 gear-set scale is 79 percent of the size of the API 613 gear-set.

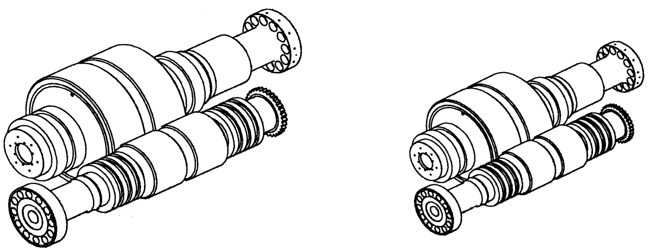

Figure 1. Illustration of Relative Sizes of API 613 Versus AGMA 6011 for Equivalent Design Conditions.

Gear elements designed based on tooth pitting index and bending stress are also checked per AGMA 6011 (1998) to ensure scuffing avoidance when provided with lubricants that do not contain extreme pressure additives. Scuffing is a damaging condition that can happen when relative movement occurs between gear tooth contact surfaces upon loss of lubricant film.

Casings

API 613 (2003) gearbox casings can be made of either cast-iron or fabricated steel but both require stress relief before final machining. In addition, casings are required to be designed to permit removing the top half without disturbing the oil piping in the bottom half of the casing.

Gear Elements and Shafts

API 613 (2003) requires pinions to be integrally forged with the shaft.

Gear wheels (bull gears) can be integrally forged with the shaft or have a one piece forged hub and rim or have a forged rim and fabricated hub and must be shrunk on the shaft with an interference fit. API 613 provides limits for each of the gear wheel fabrication options based on gear tooth pitch line velocity.

Gear wheel shafts are required to be machined from one piece of heat treated steel.

API 613 (2003) requires that a pair of mating gears have a hunting tooth combination, so a tooth on the pinion does not repeat contact with a tooth on the gear wheel until it has contacted all of the other gear wheel teeth. This is to ensure even wear on all teeth.

Each gear element is checked separately for accuracy after final machining per AGMA/ISO 1328-1 (1999).

Each pair of mating gears is checked for contact after final machining on a contact checking stand and in the job casing at the vendor's shop. This is done by applying a color transfer material such as Prussian blue at three locations 120 degrees apart to four or more teeth per location. The shafts are then rotated through the mesh, while applying a moderate drag torque, in a direction that will cause them to contact on the normally loaded faces. API 613 (2003) requires that the vendor provide the purchaser with a drawing or specification that defines acceptable contact. Unmodified gear teeth generally show a minimum of 80 percent contact across the tooth length.

Every gear under loaded operating conditions is subject to deformation of the entire rotor in three ways including bending deflection, torsional windup, and thermal distortion. Guidelines are included in API 613 (2003) for when to modify gear teeth with lead or profile modifications to provide the desired gear tooth contact under operating conditions. Contact check profile drawings are very important for gears with modified teeth to confirm that the contact pattern is consistent with the as-designed condition.

Illustrations of the effects of the loads that can create a need for lead or profile modifications along with the resulting modifications are contained in API 613 (2003) Appendix H, Figures H-1 and H-2 (attached for reference under APPENDIX B).

Gear Materials

Gear element materials are required to be forged or hot rolled alloy steel typically selected from grades identified in Appendix E of API 613 (2003). Also, material quality grades referenced to ISO 6336-5 (1996) are defined for the tooth hardening method to be applied. Guidelines for application of hardening by through-hardened, carburized, or nitrided methods are defined in API 613. A few users prefer to specify through-hardened gears, even though it may result in a larger gear unit than with other hardening methods, to provide the opportunity to increase the gear unit load rating in the future by going to carburized or nitrided gear elements.

Dynamics

A train torsional analysis of the complete coupled train including the gear unit is done by the vendor having unit responsibility. The gear unit manufacturer is required to provide the gear mass elastic data required for the torsional analysis.

A gear unit undamped lateral analysis is conducted by the gear unit manufacturer to identify the undamped critical speeds and determine their mode shapes in the range of zero percent to 125 percent of trip speed. When the specified minimum operating conditions are less than 40 percent of the gear unit power or less than 70 percent of the maximum continuous speed or the undamped analysis indicates the first critical is less than 120 percent of the maximum continuous speed, the manufacturer is required to conduct a damped unbalanced response in addition to the undamped analysis. When an unbalanced response analysis is required, an unbalance response test must be performed as part of the mechanical running test and the results used to verify the analytical model.

Pinion and gear wheel (bull gear) assemblies are required to be multiplane dynamically balanced to a maximum residual unbalance level of 4W/N.

Bearings

Radial bearings are required to be hydrodynamic bearings of the sleeve or pad type. They must be axially split and steel backed with babbitted replaceable liners, pads, or shells. Bearing liners, pads, or shells are to be mounted in axially split bearing housings and are to be replaceable without having to remove the coupling hub.

Thrust bearings are required to be steel backed multisegment type designed for equal thrust capacity in both directions and arranged for continuous pressurized lubrication to each side. Integral thrust collars are preferred per API 613 (2003) and when replaceable collars are provided they must be positively locked to the shaft to prevent fretting.

Thrust bearings and radial bearings are required to be provided with bearing temperature sensors installed as specified in API 670 (2000).

Shaft radial vibration probes and axial position probes are required along with a one event per revolution probe on the input and output shafts installed per API 670 (2000).

Lubrication

API 613 (2003) requires gear units to be pressure lubricated and provided with spray nozzles for the gear teeth. Oil systems for API 613 gear units are usually supplied by the driven equipment train manufacturer and are typically designed for ISO VG 32 mineral oil though synthetic lubricants are permitted when specified by the purchaser. When the lube oil system is supplied by the gear unit manufacturer, it is required to be designed to API 614 (1999), Chapter 2, special purpose oil system requirements.

Couplings

API 613 (2003) requires flexible couplings to be supplied by the driven equipment manufacturer per API 671 (1998) unless otherwise specified by the purchaser. Integral flange hubs are required as an API 613 default that eliminates coordination of fits of the hub to the shaft and keeps overhang low, which aids the lateral rotordynamics.

Piping

Piping is usually limited on API 613 gear units, but external piping that is provided defaults to the requirements defined in API 614 (1999).

Instrumentation

Instrumentation requirements default to API 670 (2000) for vibration and temperature measurement and monitoring and to API 614 (1999) for other types of commonly applied instrumentation.

Testing

A four hour no load at-speed mechanical run test is required as a minimum for all API 613 gear units.

Optional purchaser specified tests include:

- *Full speed/full or part load test*—Most manufacturers are limited to a part load test on larger gear units by shop driver size limitations.
- *Full torque/reduced speed test*—When a full speed full load test cannot be performed, a full torque reduced speed test may be performed to demonstrate loaded tooth contact patterns.
- *Full torque/static test*—This test would normally only be of value to check the effects of lead modification and then only if the lead modification was for reasons other than thermal distortion, since the test would not get the gear teeth to operating temperatures.
- *Back-to-back locked torque test*—This test involves two contract gear units with one serving as the test unit and the other as a drive or slave unit. It is typically done for new or modified product lines where fine tuning of lead modifications is necessary to get the results desired.
- *Sound level test*—Sound level tests are usually done for all at-speed testing, but meaningful data that reflect expectations at site conditions is difficult to achieve unless the gear unit is fully loaded on test.

APPENDIX A—

Appendix J is reproduced here courtesy of API.

APPENDIX J—RATING COMPARISON API 613 VS. AGMA 2101

J.1 General

J.1.1 This appendix describes how the rating formulas in ANSI/AGMA 2101-C95 are related to the rating methods in API Std 613 fifth edition. For definitions of terms refer to these two standards.

J.2 Tooth Pitting Index

J.2.1 Equation 27 from ANSI/AGMA 2101-C95 for calculating pitting resistance power rating is shown as Equation J1 below.

$$P_{azu} = \frac{\omega_1 b}{1.91 \times 10^7} \frac{Z_1}{K_V K_S K_H Z_R} \left(\frac{d_{w1} \sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \qquad (J1)$$

J.2.2 There are terms in AGMA that do not precisely match API terms. They are:

AGMA		API	
u	=	R	Gear ratio
ω_1	=	N_p	Pinion speed
P_a	=	Pg	Gear rated power
P_{azu}	=	$SF \cdot P_g$	Mechanical rating
F_t	=	W_t	Tangential load
b	=	F_w	Face width
d_{w1}	=	d	Pinion pitch diameter

J.2.3 Equation J2 is a conservative and reasonably accurate approximation to the more complex method found ANSI/AGMA 2101-C95 for calculating the pitting geometry factor, Z_I. The API term for ratio, R, is used.

$$Z_1 = 0.225 \frac{R}{R+1} \qquad (J2)$$

J.2.4 Using the simplified formula for Z_I, the API 613 terms, and rearranging Equation J1 algebraically results in Equation J3.

$$SFP_g = \frac{N_p d^2 F_w}{1.91 \times 10^7} \frac{0.225}{K_V K_S K_H Z_R} \left(\frac{R}{R+1} \right) \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \qquad (J3)$$

J.2.5 The tangential load is related to power and speed as shown by Equation J4.

$$W_t = \frac{1.91 \times 10^7 P_g}{N_p d} \qquad (J4)$$

J.2.6 Substituting tangential load, W_t, into Equation J3 and further rearranging gives.

$$\frac{W_t}{dF_w} \left(\frac{R+1}{R} \right) = \frac{0.225}{SF K_V K_S K_H Z_R} \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \qquad (J5)$$

Note: The terms to the left of the equal sign in Equation J5 is the K factor as defined in API Std 613 and the terms to the right is the material index number, I_m, divided by the service factor.

$$K = \frac{W_t}{dF_w} \left(\frac{R+1}{R} \right) \qquad (J6)$$

$$\frac{I_m}{SF} = \frac{0.225}{SF K_V K_S K_H Z_R} \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \qquad (J7)$$

J.2.7 The service factor, SF, is applied to the material index number, I_m, to give an allowable factor, K_a.

$$K_a = \frac{I_m}{SF} \qquad (J8)$$

J.2.8 ANSI/AGMA 2101-C95 calculates rating factors based on specific design considerations. However, API Std 613 has chosen to use conservative values for these factors and fix their numerical value. API uses the following values for the pitting rating factors:

$K_v = 1.1$	$K_s = 1.0$	$K_H = 1.3$	$Z_N = 0.68$
$Z_R = 1.0$	$Z_E = 190$	$Z_W = 1.0$	$Y_\theta = 1.0$

Note: In Equation J7 the product of the rating factors is a constant 2.0154×10^{-6} as shown in Equations J9, J10, and J11.

$$I_m = \frac{0.225}{K_V K_S K_H Z_R} \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \qquad (J9)$$

$$I_m = \frac{0.225}{1.1 \cdot 1.0 \cdot 1.3 \cdot 1.0} \left(\frac{\sigma_{HP} \cdot 0.68 \cdot 1.0}{190 \cdot 1.0} \right)^2 \qquad (J10)$$

$$I_m = 2.0154 \times 10^{-6} (\sigma_{HP})^2 \qquad (J11)$$

J.2.9 This constant shown in Equation J11 is multiplied by the allowable contact stress squared, $\sigma_{HP}{}^2$, and the result is the material index number, I_m, as shown in Table J-1.

Table J-1—Allowable Contact Stress Per API

Minimum Gear Hardness	API Material Index Number, I_m	API Contact Stress Number, σ_{HP}
302 HB†	1.38	827
321 HB†	1.53	872
341 HB†	1.70	918
363 HB†	1.89	969
90 HR15*	2.07	1013
58 Rc**	3.03	1227

†Through hardened *Nitrided **Carburized

J.3 Bending Stress Number

J.3.1 Equation 28 from ANSI/AGMA 2101-C95 for calculating bending strength power rating is shown as Equation J12.

$$P_{ayu} = \frac{w_1 d_{wl}}{1.91 \times 10^7 K_v} \frac{b m_t}{K_s} \frac{Y_J}{K_H K_B} \frac{\sigma_{FP} Y_N}{Y_\theta} \qquad \text{(J12)}$$

J.3.2 Additional terms that are different in AGMA versus API standards are listed below. Note AGMA's use of the transverse module and API use of the normal module. (See J.2.1 also)

AGMA		API	
m_t	=	$m_n \div \cos(\gamma)$	Module
P_{ayu}	=	$SF \cdot P_g$	Mechanical rating
β	=	γ	Helix angle
Y_J	=	J	Strength geometry factor

J.3.3 Using the API terms and rearranging Equation J12 algebraically results in Equation J13.

$$SFP_g = \frac{N_p d}{1.91 \cdot 10^7} \frac{F_w m_n J}{\cos\gamma} \frac{\sigma_{FP} Y_N}{K_V K_S K_H K_B Y_\Theta} \qquad \text{(J13)}$$

J.3.4 Substituting tangential load, W_t, into Equation J13 and further rearranging gives:

$$\left[\frac{W_t}{m_n F_w}\right](SF)\left[\frac{\cos\gamma}{J}\right]\frac{K_V K_S K_H K_B Y_\Theta}{Y_N} = \sigma_{FP} \qquad \text{(J14)}$$

J.3.5 ANSI/AGMA 2101-C95 calculates rating factors based on specific design considerations. However, API Std 613 has chosen to use conservative values for these factors and fix their numerical value. API uses the following values for the strength rating factors:

$K_v = 1.1$	$K_s = 1.0$	$K_H = 1.3$
$Y_\theta = 1.0$	$Y_N = 0.80$	$K_B = 1.0$

Note: In Equation J14, the product of the rating factors is very close to a value of 1.8.

$$1.8 \cong \frac{K_V K_S K_H K_B Y_\Theta}{Y_N} \cong \frac{1.1 \cdot 1.0 \cdot 1.3 \cdot 1.0 \cdot 1.0}{0.8} \qquad \text{(J15)}$$

J.3.6 Using the value shown in Equation J15 the terms to the left of the equal sign in Equation J14 is the bending stress number, S, as defined in API Std 613.

$$S = \left[\frac{W_t}{m_n F_w}\right](SF)\left[\frac{1.8\cos\gamma}{J}\right] \qquad \text{(J16)}$$

J.3.7 To arrive at the allowable bending stress number, S_a, API613 has applied an additional 25% reduction in bending stress number, σ_{FP}. This is done to increase the strength rating in relation to pitting resistance. While tooth surface damage can in many cases be detected before complete failure allowing time for scheduling repair, tooth bending failure usually leads to an immediate shutdown of the unit. The values used are shown in Equation J17 and Table J-2.

$$S_a = 0.75\sigma_{FP} \qquad \text{(J17)}$$

Table J-2—Allowable Bending Stress Peer API

Minimum Gear Hardness	API Allowable Bending Stress Number, S_a	PI Bending Stress Number, σ_{FP}
302 HB†	180	240
321 HB†	191	255
341 HB†	203	270
363 HB†	215	287
90 HR15*	190	253
58 Rc**	266	354

†Through hardened *Nitrided **Carburized

J.4 AGMA Allowable Stress Numbers

J.4.1 ANSI/AGMA 2101-C95 gives allowable stresses, σ_{HP} and σ_{FP}, based on a material "grade" and a better grade of material allows higher stresses. API Std 613 specifies the use of higher grade materials, but to be conservative bases allowable stress on the lowest grade. Both API and AGMA specifications have been revised since the original values were chosen so there is some divergence in the values today. Therefore, the API values are close to but not exactly equal to ANSI/AGMA 2101-C95 "Grade 1" values. As a comparison, Table J-3 shows the AGMA Grade 1 allowable stress numbers. These may be compared to the values chosen by API and shown in Table J-1 and J-2.

Table J-3—AGMA Grade 1 Allowable Stress Numbers

Minimum Gear Hardness	AGMA Contact Stress Number, σ_{HP}	AGMA Bending Stress Number, σ_{FP}
302 HB†	870	249
321 HB†	906	259
341 HB†	950	270
363 HB†	1006	282
90 HR15*	1070	—
58 Rc**	1240	380

†Through hardened *Nitrided **Carburized

APPENDIX B—

Appendix H figures are reproduced here courtesy of API.

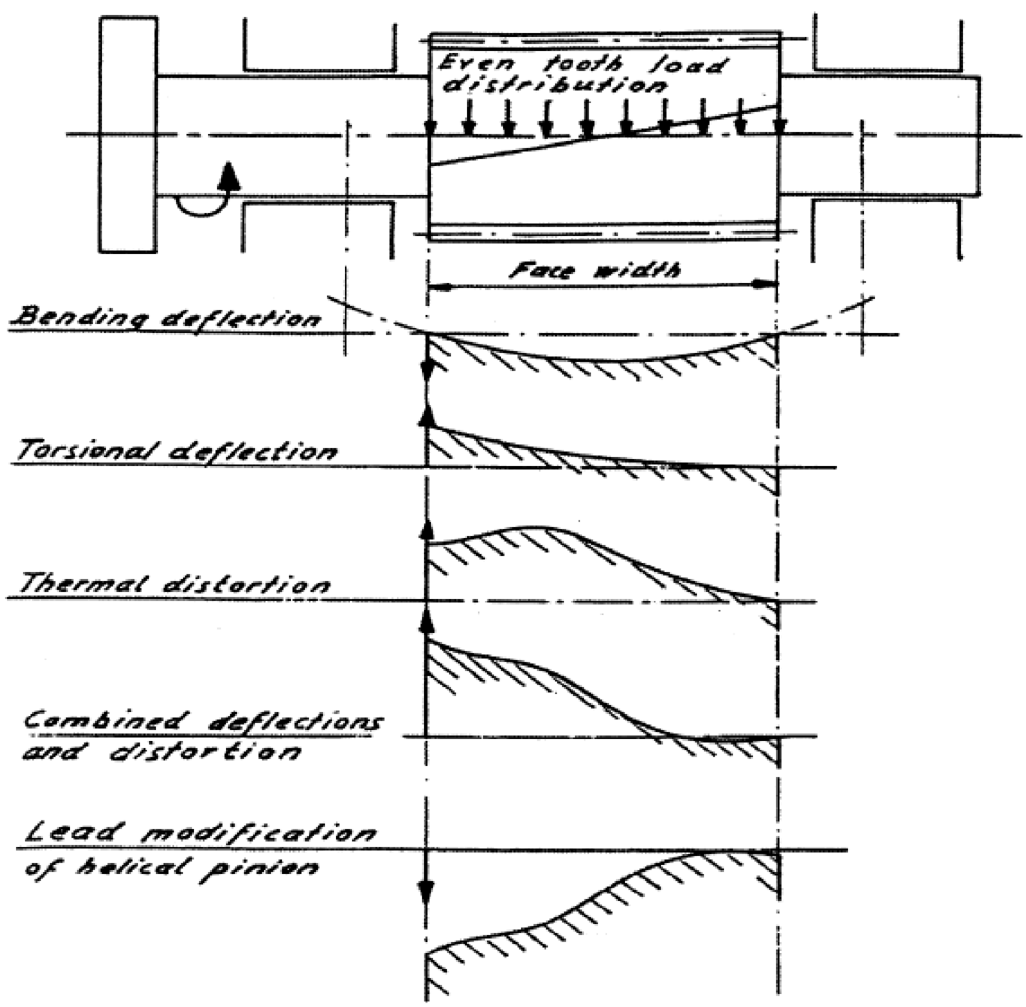

Figure H-1—Tooth Alignment (Lead) Modification of Helical Pinion

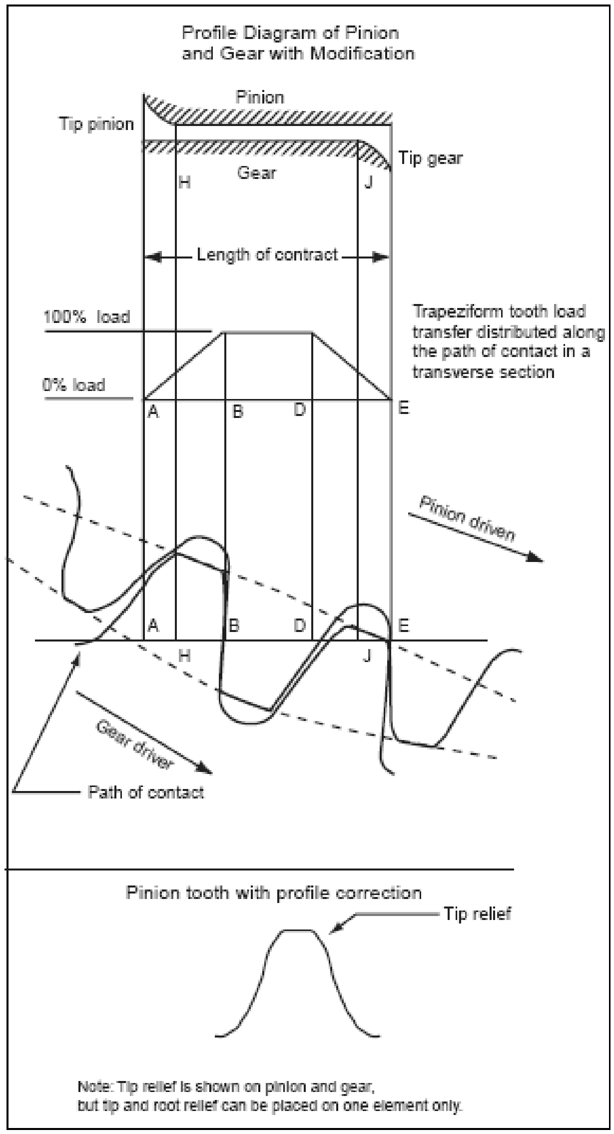

Figure H-2—Profile Modification

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